MULTIMEDIA UNIVERSITY

STUDENT ID NO

MULTIMEDIA UNIVERSITY

FINAL EXAMINATION

TRIMESTER 1, 2018/2019

EME3276 – MECHANICAL DESIGN II

(ME)

23 OCTOBER 2018
2.30 pm – 4.30 pm
(2 Hours)

INSTRUCTIONS TO STUDENTS

1. This question paper consists of 11 pages including cover page.
2. Attempt ALL questions.
3. Please write all your answers in the answer booklet provided. Show all relevant steps to obtain maximum marks.
4. You may use the appendices provided in the booklet.

Question 1

Figure Q1 shows a rotating shaft arrangement with an applied force at point *C*. Two bearings are to be mounted at point *A* and point *B* to support the shaft. An engineer recommended a Single-Row 02-20 Deep Groove ball bearing for bearing at point *A*, and a Single-Row 02-20 Angular Contact ball bearing for bearing at point *B*. The desired reliability of each bearing is 99%.

a) Find the radial load on each bearing. **[6 marks]**

b) For each bearing, determine its service life in terms of the number of revolutions, for a load application factor of 1.2. **[14 marks]**

c) Comment if the bearings will last for 15 years under operating speed of 2500 rpm. Recommend the replacement bearing(s) if necessary. **[5 marks]**

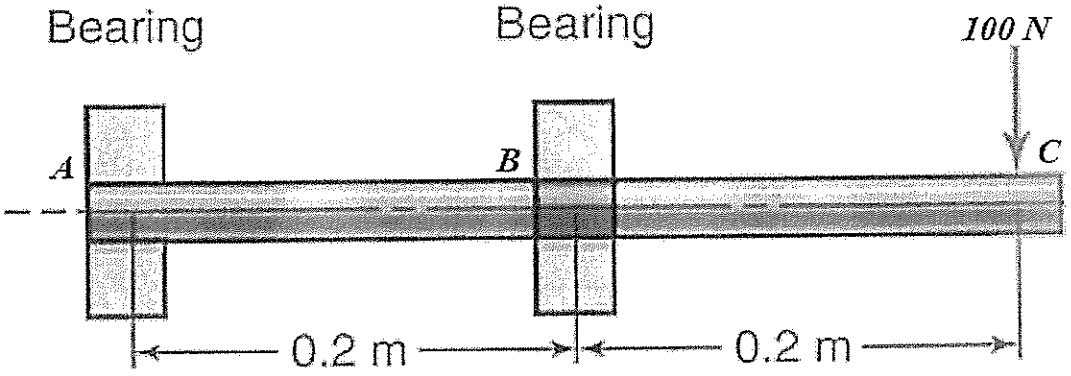

Figure Q1

Question 2

A helical-coil compression spring with plain ends is to be designed to have a maximum compression of 55 mm from its free length when subjected to a static force of 500 N. It must fit over a cylindrical rod of 30 mm outer diameter, with a diametral clearance of at least 10% of the spring coil inner diameter.

Design a suitable spring using the common spring wire and stock diameters. Ignore the need to set the spring. Use the information available in the appendix, and a design safety factor of at least 1.2 against maximum shear stress at solid length. Present your considerations of the following:

i) Wire material and size selection [Hint: 4.50 $\leq$d $\leq$5.50 mm] **[7 marks]**

ii) Maximum shear stress at solid length, taking account of curvature effect **[7 marks]**

iii) The number of active coils and the spring free length **[7 marks]**

iv) The tendency of spring to buckle, assuming the ends are hinged **[4 marks]**

Continued...

Question 3

An internal expanding centrifugal-acting rim clutch shown in **Figure Q3** has inner rim radius of 80 mm. The brake shoes have a face width of 20 mm and are both actuated by the identical force $\mathbf{F_a}$ on each shoe. The brake lining is sintered metal and has a coefficient of friction of 0.32. It has a maximum lining pressure capacity up to 1.8 MPa.

a) Calculate the actuating force $\mathbf{F_a}$. **[10 marks]**
b) Estimate the total braking capacity of the clutch.
What would happen if the left-hand shoe were turned over so as to place the hinge pin at the top? **[10 marks]**
c) Find the resulting hinge-pin reaction on the self-energizing shoe. **[5 marks]**

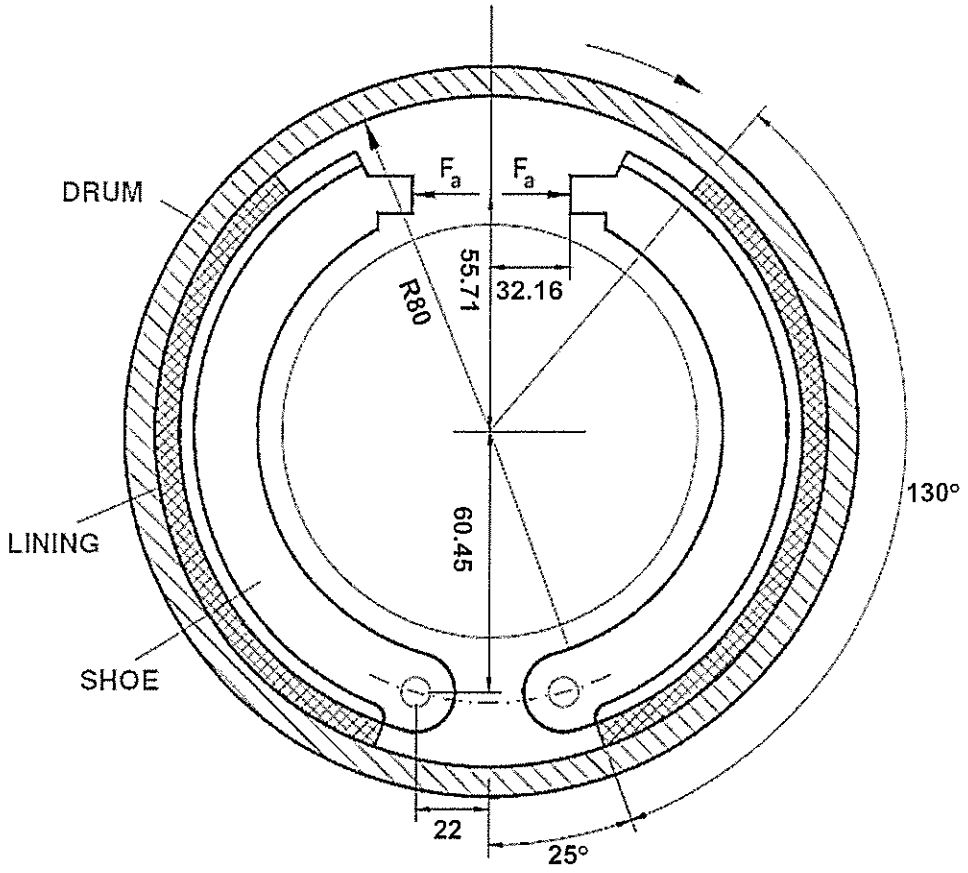

Figure Q3
[All dimensions in mm]

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Question 4

a) With the aid of a free-body diagram for a typical band brake, prove that the relationship between the tight side (P_1) and the loose (P_2) side of the band brake can be expressed as **[11 marks]**

$$\frac{P_1}{P_2} = e^{f\phi}$$

b) A band brake is designed as shown in **Figure Q4(b).** For the coefficient of friction used, the ratio between the band tensions is 1.75. The belts are fastened normal to the operating lever. A maximum force F of 600 N is applied at the end of the lever. Given $b = 40$ mm.

i) Sketch the free-body diagram of the brake. **[3 marks]**

ii) Find the coefficient of friction of the brake lining **[2 marks]**

iii) Find the band tensions if the maximum lining pressure $\mathbf{p_{max}}$ is 0.5 MPa. **[4 marks]**

iv) What must be the distance a in order that the bands be fastened normal to the operating lever? **[3 marks]**

v) Calculate the torque braking capacity. **[2 marks]**

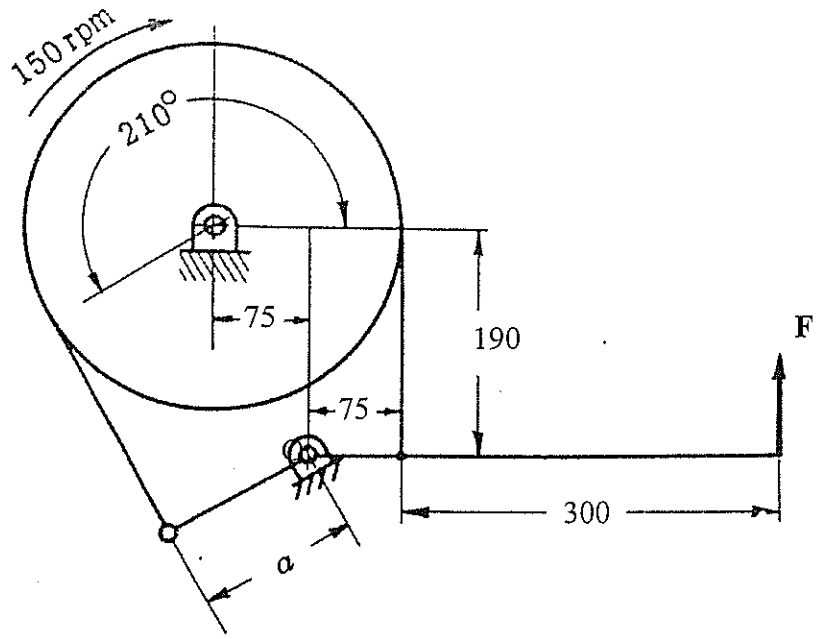

Figure Q4(b)
[All dimensions in mm]

Continued...

APPENDIX:

EQUATIONS

BEARING

$$x_D = \frac{L}{L_{10}} = \frac{60L_D n_D}{60L_R n_R}$$

$$F^a L = constant$$

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(ln1/R_D)^{\frac{1}{b}}}\right]^{\frac{1}{a}}$$

a = 3 for ball bearing and a = 10/3 for roller bearing;
b = 1.483; x_o = 0.02; $(\theta - x_o)$ = 4.439 unless otherwise mentioned

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{\frac{1}{b}}}\right]^{\frac{1}{a}}$$

if $R_D \geq 0.9$

$$F_e = X_i V F_r + Y_i F_a$$

V = 1 for inner ring rotation;
V = 1.2 for outer ring rotation

$$R = exp\left\{-\left(\frac{x_D\left(\frac{a_f F_D}{C_{10}}\right)^a - x_o}{\theta - x_o}\right)^b\right\}$$

$$F_{eq} = \left[\frac{\sum n_i t_i F_{ei}^a}{\sum n_i t_i}\right]^{\frac{1}{a}}$$

SPRING

$$k = \frac{F}{\delta} = \frac{d^4 G}{8D^3 N_a}$$

$$\tau_{max} = K_B \frac{8FD}{\pi d^3} = \frac{4C + 2}{4C - 3}\left(\frac{8FD}{\pi d^3}\right)$$

$$L_o < \frac{\pi D}{\alpha}\left[\frac{2(E - G)}{2G + E}\right]^{\frac{1}{2}}$$

$$L_o < \frac{2.63D}{\alpha}$$ for steel

$$S_{ut} = \frac{A}{d^m}$$

$$C = \frac{D}{c}$$

Continued…

CLUTCHES AND BRAKES

Internal Expanding Rim Clutches and Brakes

$$M_f = \frac{fp_a br}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin\theta(r - a\cos\theta)d\theta = \frac{fp_a br}{\sin\theta_a}\left(-r\cos\theta - \frac{a}{2}\sin^2\theta\right)_{\theta_1}^{\theta_2}$$

$$M_N = \frac{p_a bra}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin^2\theta d\theta = \frac{p_a bra}{\sin\theta_a}\left(\frac{\theta}{2} - \frac{1}{4}\sin 2\theta\right)_{\theta_1}^{\theta_2}$$

$$T = \frac{fp_a br^2}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin\theta d\theta = \frac{fp_a br^2}{\sin\theta_a}(\cos\theta_1 - \cos\theta_2)$$

$$A = \int_{\theta_1}^{\theta_2}\sin\theta\cos\theta d\theta = \left(\frac{1}{2}\sin^2\theta\right)_{\theta_1}^{\theta_2}$$

$$B = \int_{\theta_1}^{\theta_2}\sin^2\theta d\theta = \left(\frac{\theta}{2} - \frac{1}{4}\sin 2\theta\right)_{\theta_1}^{\theta_2}$$

For clockwise rotation:

$$F = \frac{M_N - M_f}{c}$$

$$R_x = \frac{p_a br}{\sin\theta_a}(A - fB) - F_x$$

$$R_y = \frac{p_a br}{\sin\theta_a}(B + fA) - F_y$$

For counterclockwise rotation:

$$F = \frac{M_N + M_f}{c}$$

$$R_x = \frac{p_a br}{\sin\theta_a}(A + fB) - F_x$$

$$R_y = \frac{p_a br}{\sin\theta_a}(B - fA) - F_y$$

Band Type Clutches and Brakes

$$\frac{P_1}{P_2} = e^{f\phi}$$

$$P_1 = p_{\max} br$$

$$T = (P_1 - P_2)\frac{D}{2}$$

Continued.....

TABLES

BEARING

Table 1: Equivalent Radial Load Factors for Ball Bearings

		$F_a/(VF_r) \le e$		$F_a/(VF_r) > e$	
F_a/C_0	e	X_1	Y_1	X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_o < 0.014$

Table 2: Dimensions and Load Ratings for Single-Row 02-Series Deep Groove and Angular-Contact Ball Bearings

			Fillet	Shoulder Diameter, mm		Load Ratings, kN			
Bore, mm	OD, mm	Width, mm	Radius, mm			Deep Groove		Angular Contact	
				d_s	d_B	C_{10}	C_0	C_{10}	C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5
75	130	25	1.5	86	119	66.3	40.5	71.5	49.0
80	140	26	2.0	93	127	70.2	45.0	80.6	55.0
85	150	28	2.0	99	136	83.2	53.0	90.4	63.0
90	160	30	2.0	104	146	95.6	62.0	106	73.5
95	170	32	2.0	110	156	108	69.5	121	85.0

Continued…

Table 3: Bearing-Life Recommendations for Various Classes of Machinery

Type of Application	Life, k_h
Instruments and apparatus for infrequent use	Up to 0.5
Aircraft engines	0.5–2
Machines for short or intermittent operation where service interruption is of minor importance	4–8
Machines for intermittent service where reliable operation is of great importance	8–14
Machines for 8-h service that are not always fully utilized	14–20
Machines for 8-h service that are fully utilized	20–30
Machines for continuous 24-h service	50–60
Machines for continuous 24-h service where reliability is of extreme importance	100–200

Table 4: Load-Application Factors

Type of Application	Load Factor
Precision gearing	1.0–1.1
Commercial gearing	1.1–1.3
Applications with poor bearing seals	1.2
Machinery with no impact	1.0–1.2
Machinery with light impact	1.2–1.5
Machinery with moderate impact	1.5–3.0

Table 5: Formulas for the Dimensional Characteristics of Compression-Springs

	Type of Spring Ends			
Term	Plain	Plain and Ground	Squared or Closed	Squared and Ground
End coils, N_e	0	1	2	2
Total coils, N_t	N_a	$N_a + 1$	$N_a + 2$	$N_a + 2$
Free length, L_0	$pN_a + d$	$p(N_a + 1)$	$pN_a + 3d$	$pN_a + 2d$
Solid length, L_s	$d(N_t + 1)$	dN_t	$d(N_t + 1)$	dN_t
Pitch, p	$(L_0 - d)/N_a$	$L_0/(N_a + 1)$	$(L_0 - 3d)/N_a$	$(L_0 - 2d)/N_a$

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Table 6: End-Condition Constants α for Helical Compression Springs

End Condition	Constant α
Spring supported between flat parallel surfaces (fixed ends)	0.5
One end supported by flat surface perpendicular to spring axis (fixed); other end pivoted (hinged)	0.707
Both ends pivoted (hinged)	1
One end clamped; other end free	2

*Ends supported by flat surfaces must be squared and ground.

Table 7: Constants A and m of $S_{ut} = A/d^m$ for Estimating Minimum Tensile Strength of Common Spring Wires

Material	ASTM No.	Exponent m	Diameter, in	A, kpsi · in^m	Diameter, mm	A, MPa · mm^m	Relative cost of Wire
Music wire*	A228	0.145	0.004–0.256	201	0.10–6.5	2211	2.6
OQ&T wire*	A229	0.187	0.020–0.500	147	0.5–12.7	1855	1.3
Hard-drawn wire†	A227	0.190	0.028–0.500	140	0.7–12.7	1783	1.0
Chrome-vanadium wire‡	A232	0.168	0.032–0.437	169	0.8–11.1	2005	3.1
Chrome-silicon wire‖	A401	0.108	0.063–0.375	202	1.6–9.5	1974	4.0
302 Stainless wire§	A313	0.146	0.013–0.10	169	0.3–2.5	1867	7.6–11
		0.263	0.10–0.20	128	2.5–5	2065	
		0.478	0.20–0.40	90	5–10	2911	
Phosphor-bronze wire¶#	B159	0	0.004–0.022	145	0.1–0.6	1000	8.0
		0.028	0.022–0.075	121	0.6–2	913	
		0.064	0.075–0.30	110	2–7.5	932	

Table 8: Maximum Allowable Torsional Stresses for Helical Spring in Static Applications

Material	Maximum Percent of Tensile Strength	
	Before Set Removed (Includes K_W or K_B)	After Set Removed (Includes K_S)
Music wire and cold-drawn carbon steel	45	60–70
Hardened and tempered carbon and low-alloy steel	50	65–75
Austenitic stainless steels	35	55–65
Nonferrous alloys	35	55–65

Continued...

Table 9: Mechanical Properties of Some Spring Wires

Material	Elastic Limit, Percent of S_{ut} Tension	Torsion	Diameter d, in	E Mpsi	E GPa	G Mpsi	G GPa
Music wire A228	65–75	45–60	<0.032	29.5	203.4	12.0	82.7
			0.033–0.063	29.0	200	11.85	81.7
			0.064–0.125	28.5	196.5	11.75	81.0
			>0.125	28.0	193	11.6	80.0
HD spring A227	60–70	45–55	<0.032	28.8	198.6	11.7	80.7
			0.033–0.063	28.7	197.9	11.6	80.0
			0.064–0.125	28.6	197.2	11.5	79.3
			>0.125	28.5	196.5	11.4	78.6
Oil tempered A239	85–90	45–50		28.5	196.5	11.2	77.2
Valve spring A230	85–90	50–60		29.5	203.4	11.2	77.2
Stainless steel							
A313*	65–75	45–55		28	193	10	69.0
17-7PH	75–80	55–60		29.5	208.4	11	75.8
414	65–70	42–55		29	200	11.2	77.2
420	65–75	45–55		29	200	11.2	77.2
431	72–76	50–55		30	206	11.5	79.3

Table 10: Commonly Available Spring Wire Stock Diameters

in	mm	in	mm
0.010	0.25	0.092	
0.012	0.30	0.098	2.50
0.014	0.35	0.105	
0.016	0.40	0.112	2.80
0.018	0.45	0.125	
0.020	0.50	0.135	3.50
0.022	0.55	0.148	
0.024	0.60	0.162	4.00
0.026	0.65	0.177	4.50
0.028	0.70	0.192	5.00
0.030	0.80	0.207	5.50
0.035	0.90	0.225	6.00
0.038	1.00	0.250	6.50
0.042	1.10	0.281	7.00
0.045		0.312	8.00
0.048	1.20	0.343	9.00
0.051		0.362	
0.055	1.40	0.375	
0.059		0.406	10.0
0.063	1.60	0.437	11.0
0.067		0.469	12.0
0.072	1.80	0.500	13.0
0.076		0.531	14.0
0.081	2.00	0.562	15.0
0.085	2.20	0.625	16.0

[1]Custom wire sizes also available at extra cost.

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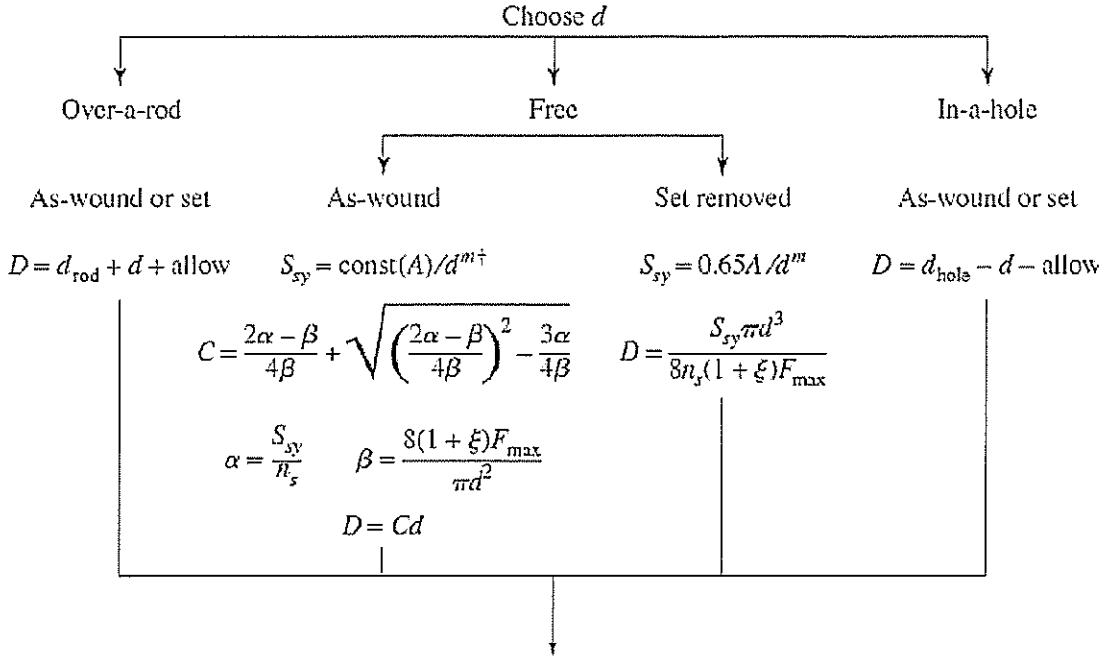
FIGURES

SPRING

Figure 1: Design Flowchart for Static Loading



$$C = D/d$$

$$K_B = (4C + 2)/(4C - 3)$$

$$\tau_s = K_B 8(1 + \xi)F_{max}D/(\pi d^3)$$

$$n_s = S_{sy}/\tau_s$$

$$\text{OD} = D + d$$

$$\text{ID} = D - d$$

$$N_a = Gd^4 y_{max}/(8D^3 F_{max})$$

$$N_t$$

$$L_s$$

$$L_O$$

$$(L_O)_{cr} = 2.63D/\alpha$$

$$\text{fom} = -(\text{rel. cost})\gamma\pi^2 d^2 N_t D/4$$

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